A

PROJECT REPORT

ON

RE-CHARGING OF A RECTANGULAR MICROCHANNEL OVER A FIXED SPAN TO INCREASE HEAT TRANSFER

BY

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UNDER THE GUIDANCE OF

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CERTIFICATE

This is to certify that the thesis entitled "**Recharging of a rectangular microchannel over a span to increase heat transfer**" Submitted by **Mr. K ABHINAV (111ME0278)** in partial fulfillment of the requirements for the merit of **Bachelor of Technology** Degree in Mechanical Engineering at National Institute of Technology Rourkela is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in this thesis has not been submitted to another university/ institute for award of any Degree or Diploma.

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SELF DECLARATION

I, Mr. K. Abhinav, Roll No. 111ME0278, student of B. Tech (2011-15), at Department of Mechanical Engineering, National Institute of Technology Rourkela do hereby declare that I have not adopted any kind of unfair means and carried out the research work ethically to the best of my knowledge. If adoption of any kind of unfair means is found in this thesis work at a later stage, then appropriate action can be taken against me including withdrawal of this thesis work.

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26 May 2015

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I am extremely fortunate to be involved in an exciting and challenging research project like "Recharging of a rectangular microchannel over a span to increase heat transfer". It has enriched my life, giving me an opportunity to work in a new environment of ANSYS (FLUENT). This project increased my thinking and understanding capability as I started the project from scratch.

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ABSTRACT

Microchannel have gained immense popularity due to its extensive use in various fields, especially for heat exchangers used to cool the microelectronic chips and also for various other purposes. This project focuses on increasing heat transfer on the conjugate wall by carrying out a mathematical simulation to study the effect on the Nusselt number by the introduction of a fresh flow by a secondary recharging inlet in between the main channel flow. At first a 2-D model representation (i.e. flow between two plates) has been studied to check the viability of the recharging concept through comparison between a simple and recharging case. Proceeding with the 3-D case, numerical analysis has been carried out for three varying geometries ($\delta_{sf} =$ 1, 2, 4) of microchannel with two Reynold's number (Re = 500, 1000) variation and 11 different solid materials (k_{sf} = 0.33~702.9) with variable thermal conductivity values. A constant heat flux has been provided at the bottom wall, other walls being kept adiabatic and flow rate is assumed such that the flow is laminar throughout theoretically. It is found that the introduction of a recharging inlet causes an increase in the average Nusselt number which represents an increase in heat transfer thus refining the efficiency of the microchannel.

Further the effect of axial conduction on the local Nusselt number is studied by varying k_{sf} values i.e. by changing material. It is found out that too high a value causes more back conduction thus decreasing Nusselt number. Whereas too low value of thermal conductivity also decreases the Nusselt number by altering the spatial distribution of the heat. Thus optimum k_{sf} for maximizing Nusselt number is found out in between the two extreme values by plotting the graphs. Added to these the effects of varying geometry on flux and temperature distribution has also been analyzed. Hence, it is seen that, recharging increases the average Nusselt number and also k_{sf} is the key value in deciding the effect of axial condition and thus maximizing the Nusselt number.

Keywords: microchannel, recharging, axial back conduction, laminar flow.

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NOMENCLATURE

A_c	Cross-sectional area, m^2
C_p	Specific heat capacity of fluid, J/kg K
D_h	Hydraulic diameter, m
h	Convective heat transfer coefficient, W/m K
k_f	Thermal conductivity of fluid, W/m^2 K
k_s	Thermal conductivity of solid, W/m^2 K
k_{sf}	Thermal conductivity ratio of solid to fluid
L	Length of the microchannel, m
m	Average mass flow rate, m/s
Nu_z	Local Nusselt number
Nu_{avg}	Average Nusselt number
P	Wetted Perimeter, m
P_1, P_2	Pressure at sections 1 and 2 , Pa
Pr	Prandtl number
q'	Heat flux at the conjugate wall, W/m^2
q	Applied heat flux, W/m^2
Re	Reynold's number
T_f	Bulk fluid temperature, K
T_w	Wall temperature, K
V	Velocity of fluid, m/s
Z	Axial distance in z direction, mm
α	Thermal diffusivity,
δ	Thickness, mm
δ_f	Channel width of the fluid domain, mm
δ_s	Thickness of the solid substrate, mm
δ_{sf}	Ratio of δ_s and δ_f (δ_s/δ_f)
λ	Conduction parameter

μ	Dynamic viscosity, Nm/s
ν	Kinematic viscosity, m^2/s
ϕ	Dimensionless flux ratio
ρ	Density, kg/m^3
Θ	Dimensionless temperature ratio, -

Chapter 1
Introduction

1.1 Microchannel and its usage in different fields

In the most recent couple of decades, inferable from the advancements in technology especially in field of miniaturized manufacturing, the electronic, biomedical and aerospace fields are revolutionized by the micro devices. The most widely used devices being micro biochips, micro motors, and micro heat-exchangers. Because of this miniaturization of devices especially in electronic field, overheating has been the main problem to be countered. To provide a solution for high flux cooling problems the use of convection heat transfer in microchannel is believed to be the most feasible and logical way.

Geometrically microchannel refers to those channels that have hydraulic dimensions less than 1 mm and greater than 1 micron [21]. Channels above 1 mm shows same characteristic flow as the conventional channels so they are termed as macro channels, whereas channels with dimensions below 1 micron are coined as nanochannels. Microchannel gives a focal point of high surface to volume proportion. As the volume declines, surface territory diminishes, however the amount of fall in value of these two quantities is diverse. Volume diminishes more than surface territory and consequently microchannel give the high surface region to volume proportion. Because of high surface territory to volume proportion of microchannel, heat exchange coefficient increases. Because of these properties, microchannel is generally utilized as a part of electronic devices, micro heat exchangers and so on.

In microchannels, the ratio of hydraulic diameter to the solid substrate wall thickness is one whereas in macro channels the ratio is less than unity. The application of liquid flow in microchannels for cooling has spiked up since last 25 years. Owing to the extensive experimentation on convective heat transfer, the flow regimes has been studied and found to be in clash with some of conventionally formulated channel flows.

Turkerman and Pease [1] has depicted that microchannel are equipped for extricating expansive measure of heat, and in this manner act as heat sink. Consequently they are extensively used in IC's, transistors and so on. Heat extraction from any surface is capacity of particular heat limit, surface area and temperature distinction of wall and flowing liquid. For electronic devices, temperature are confined as far as possible, thus more accentuation is done on surface range and temperature distinction of wall and liquid. Because of higher surface zone/volume when contrasted with ordinary channel, microchannel gives high heat extraction.

Thus in this concerned thesis we are going to evaluate the effect of recharging that could help increasing the heat transfer in microchannel thus enhancing its optimized applicability in various fields.

1.2 Introduction to internal pipe flow

Due to its potentially extensive use in the field of heat exchangers involved in reactor, miniature circuits, it is essential to study its heat transfer characteristics so as to increase its efficiency. The flows in pipes or channels commonly used in cooling purposes are essentially case of forced convection as the flow is forced by a pressure difference. In case of internal flow in pipes the boundary has a limit up to which it can grow unlike in external flow. Flow in a tube can be laminar or turbulent depending on flow conditions. At low velocity fluid flow is streamlined and laminar. For flow in circular tubes (most popularly used as for a given surface area gives the ultimate heat transfer owing to the least pressure drop), the Reynolds number is defined as

$$\mathrm{Re} = \frac{\rho V D}{\mu} = \frac{VD}{\nu} \tag{1.1}$$

where, V = mean velocity of fluid (m/s), D = diameter of tube (m), μ = dynamic viscosity (m^2/s). For the noncircular channels as in square or rectangular, the same formula is used to calculate Reynolds number by defining a term hydraulic diameter,

$$D_h = 4A_c/P \tag{1.2}$$

where A_c = cross-sectional area (m^2), P = wetted perimeter of the cross-section (m). The value of Reynolds number below 2300 ensures a laminar flow. So for this analysis of microchannel the Reynolds number is kept below 2300.

As we know the continuum assumption necessarily validates the no-slip condition at the conjugate wall, i.e. developing of a boundary layer, to make up for velocity reduction, the velocity at centerline increases. The region from tube inlet to the phase where the boundary layer merges with the centerline is called the hydrodynamic entrance region (L_h). After this region the flow is called fully developed flow. The same concept also arises considering thermal boundary layer that gives rise to two sections thermally developing flow and thermally fully developed flow as shown in Fig. 1.1.

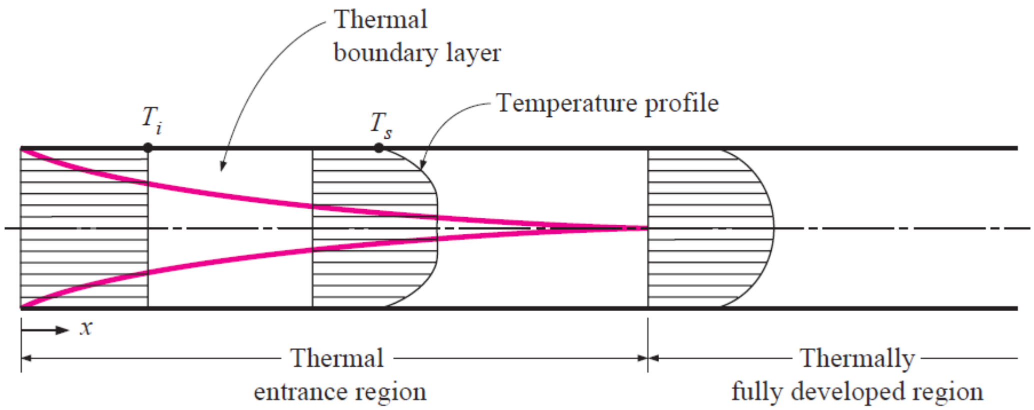

Fig. 1.1. Development of thermally fluid developed region in a channel (the liquid in tube is being cooled) [5].

Consider a fluid that is being heated (or cooled) in a tube as it flows through it. The friction factor and the heat transfer coefficient are highest at the tube inlet where the thickness of the boundary layers is zero, and decrease gradually to the fully developed values, as shown in Fig. 1.2 from Cengel [5]. Therefore, the pressure drop and heat flux are higher in the entrance regions of a tube, and the effect of the entrance region is always to enhance the average friction and heat transfer coefficients for the entire tube. This enhancement can be significant for short tubes but negligible for long ones.

The variation in heat transfer coeffiecient is reflected in the value of Nusselt number as can be infered from the formula. So in order to increase the average nusselt number the developing region has the most to contribute. The local Nusselt number for a channel is given by

$Nu_{avg} = hD_h/k$ (1.3)

where, k = thermal conductivity of fluid, W/m^2-K

The fluid flowing channel can again be either have two thermal conditions subjected to analysing that are: (a) constant heat flux, (b) constant surface temperature on its outer surface. The present analysis of the microchannel is done considering a constant heat flux on the outer wall. Thus the Nusselt number, flux and temperature variations are studied in this case.

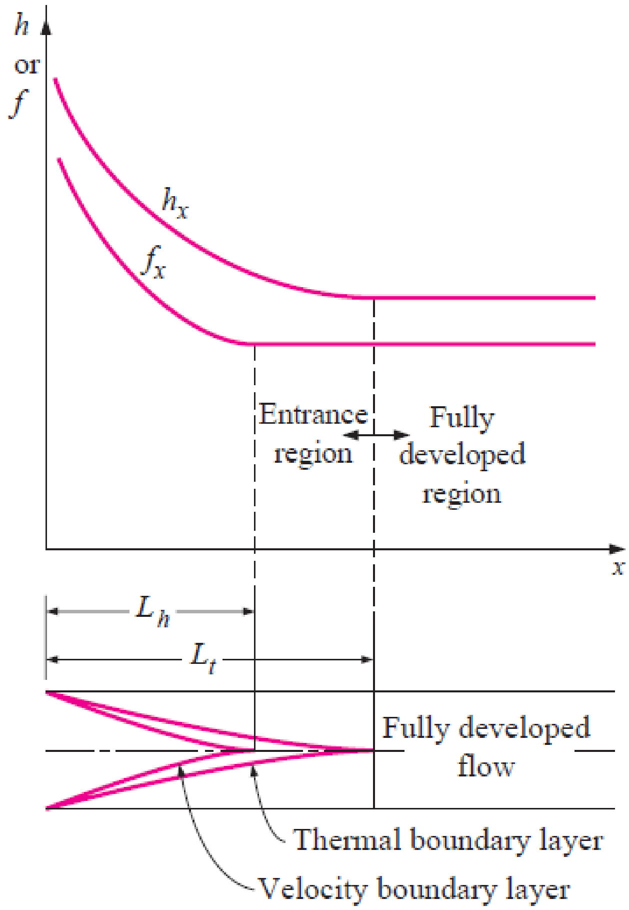

Fig. 1.2. Variation of friction factor and convection heat transfer coefficient along the channel length [5].

1.3 Axial back conduction

In conventional channels or tubes or ducts, the solid substrate thickness is smaller compared to the hydraulic diameter of pipe i.e. the ratio of the solid thickness (δ_s) to the hydraulic diameter is less than unity. This ratio is represnted by δ_{sf} (or δ_s/δ_f). The discussion of this ratio is essential to understand the phenomenon of axial back conduction, which is conveniently neglegible in conventional channels.

Basically the axial back conduction in wall decreases the heat carrying capacity of fluid due to back flow of energy. Several studies has been done to study the effect of axial conduction in walls of microchannels. To numerically represent the effect of axial conduction in the substrate on internal single-phase convective flows, the concept of "conduction parameter," a quantity that gives relative importance of conduction heat transfer compared to the energy flow carried by the fluid exists. It can be defined as the ratio of axial heat transfer within the solid channel or tube due to axial temperature gradient in it to the energy flow carried by the fluid in the channel in the axial direction. This parameter was used by Bahnke and Howard [2] for simulation of recuperators, and in later years, by Peterson [3,4] to study conduction effect in microscale counter-flow heat exchangers. It is defined as

$$\lambda = \frac{K_s A_s}{m C_p L} \tag{1.4}$$

where the average mass flow rate is m = $\rho_f \cdot A_f \cdot u$. The physical understanding of the impacts because of axial conduction in the substrate can be concentrated on by considering the three characteristic nondimensional parameters of this issue, i.e.,

- number of exchange units of the framework, NTU,
- Biot number, defined on the premise of the thickness of the substrate
- wall size proportion.

1.4 Concept of Recharging

As inferred from the previous section, the main region that affects the heat transfer is the developing region in a flow, the region with developing boundary layer. A higher heat transfer rate would result in higher heat absorption by the fluid thus giving an efficient cooling or waste heat extraction system. Now this could be seen as a perspective of increasing the developing region of fluid that would affect the average heat transfer values. The increase in developing region could be achieved by introducing a fresh flow in the system at regular intervals across the channel length. This concept is termed as recharging of a channel as the study is focused on microchannel this is referred as recharging of microchannel. The expected change in Nusselt number graph for recharging channel from those of conventional is likely to be somewhat as shown below.

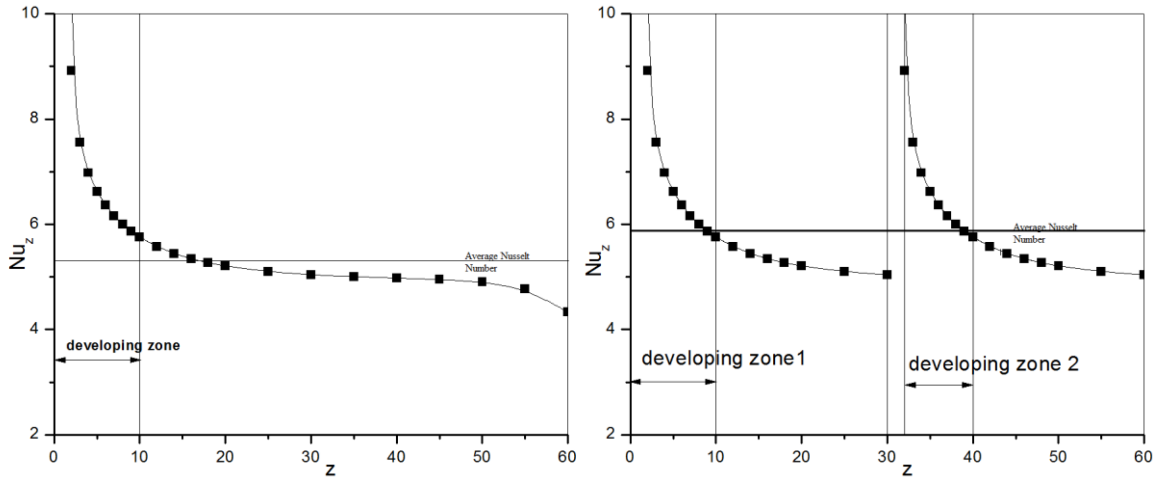

Fig. 1.3. Comparison of local Nusselt number in (a) a conventional microchannel of length 60 mm, and (b) a recharging microchannel where each part of length of 30 mm.

As developing region increases, average heat transfer coefficient increases leading an average increase in heat transfer due to absence of boundary layer in these regions. Thus, the average heat transfer should increase for the same span of fluid flow making it more efficient.

The recharging is to be done by introducing a outlet and inlet at mid-span or at regular intervals (if more than one recharging is to be done). At the recharging outlet , it is subjected to a pressure such that the fluid diverges into the outlet instead of continuing the flow and also the pressure should not cause any back flow creating turbulence in the small region between the inlet and outlet. Theoretically this pressure could be calculated from the Hagen-Poiseuille Formula

$$P_1 - P_2 = \frac{32\mu \upsilon L}{D^2} \tag{1.5}$$

μ = viscosity of fluid (N-s/m^2), v = average velocity (m/s), L = length from inlet (m), D = hydraulic diameter (m).

But for the sake of this project the pressure calculation is done by running the simulation for a half channel due to symmetry.

In spite of the fact that various numerical and test studies have been performed on the impact of wall heat conduction on hydrodynamically and/or thermally developing flows in mini/ microchannel and tubes, an express recognizing parameter for judging the impact of axial conduction in the substrate on the local and average Nusselt number in the passage length is not accessible. Sufficient number of parametric studies is likewise not promptly accessible; most reported studies manage just round microtubes. Rectangular microchannels are exceptionally compelling as they are utilized widely as warmth sinks as a part of microelectronic gadgets, and for reactant reactors for smaller scale fuel processors, organic sensors, and so on. Such rectangular small scale/microchannels are frequently utilized in high heat flux disseminating heat exchanger equipment. So the considered case is also rectangular channel for a more beneficial analyzing of the case.

Chapter 2
Literature Review

Microchannel study is not a new concept, rather a well developed and experimented field in heat transfer community. A review on microchannel and axial heat conduction is studied some being experimental other being CFD analysis.

Tullius et al. [6] have analyzed mini-channels with fins for maximization of heat transfer by convection at conjugate walls. Six various shapes – triangle, circle, ellipse, diamond, square and hexagon has been modelled with a constant heat flux source at bottom. They concluded that triangular shape provides high heat transfer due to increase in Nusselt number. As the results inferred Nusselt number can also be increased by increasing fin height, decreasing fin width and spacing.

Jasperson et al. [7] have done a numerical analysis to compare the heat transfer rates on micro and macro channel and it is found that microchannels are more efficient as a heat sink. Further proceeding finned microchannels showed potential to enhance heat transfer and may be replace the conventional channels. Thermal and hydraulic performance along with the cost of manufacturing as metrics are also compared.

Qu and Mudawar [8] studied a 3-D rectangular microchannel fluid flow and analyzed numerical solution using water as coolant. Finite difference method based numerical coding and SIMPLE algorithm was used to solve conservation equations. It is found that the temperature rise in both solid and fluid era follows a linear increase with the highest temperature at outlet. Heat flux and Nusselt number has highest value at inlet.

Qu and Mudawar [9] analyzed to check the variation of pressure drop and heat transfer characteristics along the microchannel heat sink, the study is carried out numerically as well as experimentally. De-ionized water is used as coolant fluid and microchannel is constructed by oxygen free copper. Analysis confirms the good agreement of experimental and theoretical data thus validating the Navier-Stokes and energy equation for microchannel flow.

Ho et al. [10] carried out experimental work to study the pressure drop and heat transfer characteristics for a finned microchannel heat sink. Reynold s number were varied. It is concluded from experiments that friction factor calculated from experimentation matched well with Moore and Joshi's theoretical values of friction factor coefficient. Higher local heat transfer coefficient is found only at inlet.

Guo and Li [11] studied a microchannel and analyzed the size effect on flow and heat exttraction. Gas was taken as the fluid medium that's why Knudsen number. was calculated and maintain near to 0.001 to maintain the continuum assumptions. Since microchannel has large surface area to volume ratio, the surface effects employ a greater importance. As this ratio increases, the induced compressebilty flatters the velocity profile and enhances the friction factor and Nusselt number.

Lee et al. [12] carried out an experimental way to study heat transfer and modelled a rectangular channel. Fluid flowing is deionised water. The study was done using a continuum approach to check on the microchannel being out of statistical approach. On the other hand numerical simulations were carried out using Navier-Stoke's equation. And both the result shows a good convergence with each other.

Hestroni et al. [13] analysed microchannel using both gas and liquid as cooling fluids at low Knudsen number and high Mach number.Variation of Reynold's number is accounted to study the transformatio of flow from laminar to turbulent flow. And checked the similarity of conventional methods with experimental results.

Morini [14] studied viscous heating in during the flow in microchannel. He showed the effect of cross section on viscous heating. And described the role of Reynolds number on flow heating. He formulated for minimum Reynolds number at which viscous dissipation can be overlooked. Formula is based on the parameters hydraulic diameter and geometry. He also compared the experimental result with conventional theory of Navier-Stokes equation.

Lee and Garimella [15] did 3-D numerical simulations to know the effect of convective heat transfer in the entrance zone of laminar flow in a rectangular microchannel. Two boundary conditions were considered (a) constant wall temperature & (b) constant heat flux. Variation of aspect ratio is also considered. Both Nusselt number and average Nusselt number are calculated and plotted for constant temperature and constant heat flux. Formulae were designed which shows good consent with other experimental results.

Moharana et al. [16] performed a numerical analysis on rectangular pipe flow microchannel, to optimize the Nusselt number. They explained the effect of axial wall conduction and heat transfer with constant heat flux at bottom face of rectangular channel. They also explained effect of thermal conductivity on heat transfer. They also pointed out the dependency of substrate thickness to channel thickness.

Moharana et al. [17] carried out various numerical simulations and experimentation for thermohydrodynamic performances of single phase flow. An array of rectangular microchannel was designed for experimental work and similar for numerical simulation. Microchannel array was fabricated on copper substrate. Reynolds number varied from 150 to 2500 and prandtl number from 3 to 4. Inlet pressure is kept at 1.1 bar. Transformation of laminar to turbulent flow is found at the Reynolds number 1100 for channel roughness 3.3 μm. A 3 D model were designed to correspond to experimental geometry. In boundary condition bottom wall is given a constant heat flux. Based on the combined study they concluded that conjugate effects play vital role in mini/microchannel system and negligence of axial wall conduction leads to wrong results and conclusions.

Moharana and Khandekar [18] studied the axial conduction in microtubes. 2-D numerical simulation was carried out. Constant heat flux and constant temperature both condition were analysed at outer wall of microtube. The flow is assumed as laminar. The cross sectional faces are considered as insulated. Different material of substrate was used and ratio of thermal conductivity of solid to liquid is the main parameter. Ratio of tube thickness varied from 1 to 16. Result clearly indicated k_{sf} plays important role in the conjugate heat transfer.

Moharana and Khandekar [19] studied conjugate heat transfer in three dimensional numerical simulations. The flow is considered as laminar in rectangular microchannnel to evaluate the effect of aspect ratio on axial conduction in the substrate. A reference fixed size rectangular channel of dimensions 0.6 mm × 0.4 mm × 60 mm was used and variation in aspect ratio were done. Variation of aspect ratio follows the width to height from 0.45 to 4.0. the flow rate is low and Reynolds number is taken as 100. Simulation concluded with the result that maximum value of Nusselt number exists at the optimum value of thermal conductivity of solid to liquid.

Tiwari et al. [20] carried out study on influence of axial conduction in a partially heated microtube. In this study a numerical analysis were carried. The flow was considered as laminar and constant heat flux condition was imposed on the outer wall surface. Comparisons were made with fully heated microtube. Total length of microtube is taken as 60 mm × 6 mm from both inlet and outlet end are insulated. Cross sectional face of microtubes are insulated. Variation in three parameter considered are Reynolds number, ratio of thermal conductivity of solid to liquid and ratio of wall thickness to inner radius. They found that wall thickness and conductivity play critical role in axial back conduction.

From the above review it can be inferred that recharging of a microchannel is a new concept to be introduced which could in time gain a practical use after proper experimentation and analysis.

Chapter 3
Numerical Simulation

3.1 Introduction

A study has been done on increasing the average Nusselt number in a rectangular microchannel. In this project, a rectangular microchannel is modified by providing a secondary outlet and a secondary inlet in between the primary flow channel. The Nusseltnumber has been calculated and compared to that of a simple rectangular cross-sectional microchannel. The rectangular channel considered is of length 60 mm with the recharging section provided after 29.5 mm from the inlet with recharging outlet of dimension 0.4 mm × 0.4 mm and recharging inlet dimension as 0.4 mm × 0.4 mm with a gap of 0.2 mm in between them . For simulation purpose only the 50% of the geometry is considered. The Nusselt number has been calculated for the entire channel and compared that with a simple channel without recharging. After that, the further study includes studying the effect of varying Reynold's number, materials according to their thermal conductivities and also the geometry is being varied by varying the solid substrate width to study the axial conduction effect.

The solid models were created using commercial software SOLIDWORKS and then transferred onto commercially available ANSYS by .igs format files. The simulation is done in ANSYS FLUENT environment under ANSYS V15.0 software. In total around 70 simulations are done and the calculated values are charted in Microsoft Excel 2013.

Before proceeding to the simulation of a 3-D recharging microchannel, a 2-D simulation is tested for a similar case i.e. for flow between two plates. Here the recharging splits are provided on the upper plate and the lower plate is subjected to a constant heat flux. The variation of local Nusselt number is plotted and compared to that of a simple channel.After we get the desired result in this case. 3D-simulation work is started.

The variation considered are (a) Reynold's number: 100, 50 (b) Geometrical width ratio of hydraulic dia to substrate thickness: 1, 2, 4 (c) Material according to their conductivity by k_{sf} values.

Table 1: Material and their properties

Metal	Density (kg/m^3)	Heat Capacity (J/kg-K)	Thermal Conductivity (k_s)	Thermal Conductivity Ratio (k_{sf})
Sulfur	2070	708	0.206	0.33
SiliconDioxide	2220	745	1.38	2.26
Bismuth	9780	122	7.86	12.87
Nicrome	8400	420	12	19.66
SS 316	8238	468	13.4	21.99
Constantan	8920	384	23	37.68
ChromiumSteel	7822	444	37.7	61.77
Bronze	8780	355	54	88.47
Zinc	7140	389	116	190.06
Alloy 195	2790	883	168	275.26
Silver	10500	235	429	702.9

3.2 The differential equations used for modelling of simulation are :

(a) Continuity Equation:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho V) = 0 \tag{3.1}$$

(b) Momentum Equation:

$$\rho\left(\frac{\partial u}{\partial t} + \frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} + \frac{\partial u}{\partial z}\right) = -\frac{\partial P}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) + \mathrm{F_x} \tag{3.2}$$

$$\rho\left(\frac{\partial v}{\partial t} + \frac{\partial v}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial v}{\partial z}\right) = -\frac{\partial P}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + \mathrm{F}_y \tag{3.3}$$

$$\rho\left(\frac{\partial w}{\partial t} + \frac{\partial w}{\partial x} + \frac{\partial w}{\partial y} + \frac{\partial w}{\partial z}\right) = -\frac{\partial P}{\partial z} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) + \mathrm{F_z} \tag{3.4}$$

(c) Energy Equation:

$$u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) + \frac{1}{\rho C}\phi \tag{3.5}$$

$$\phi = 2\mu\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \frac{1}{2}\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)^2 - \frac{1}{3}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right)^2\right]$$

3.3 Assumptions in modelling of microchannel

To analyze the heat transfer characteristics in microchannel some assumption are to be made such as :

(1) No variation in density of fluid along the flow, incompressible fluid.

(2) The system is in steady state.

(3) Heat transfer by natural convection is negligible.

(4) Heat transfer by radiation in microchannel heat sink is negligible.

(5) Fluid properties remain same throughout the flow.

(6) Axial conduction effects are onsidered.

(7) The microchannel flow condition follows same equations as macrochannel flow i.e. Navier-Stoke’s equation is valid for the microchannel.

3.4 Geometrical model of the microchannel

- A metal substrate of size 60 mm × 1.2 mm × 1 mm is used and a channel of 0.4 mm × 0.4 mm is carved on it.
- Considering symmetry, only half of the portion is considered.
- A parting plane is done on the half model.
- Recharging portion is from 29.5 mm – 30.5 mm (0.4 mm outlet + 0.2 mm gap + 0.4 mm inlet)

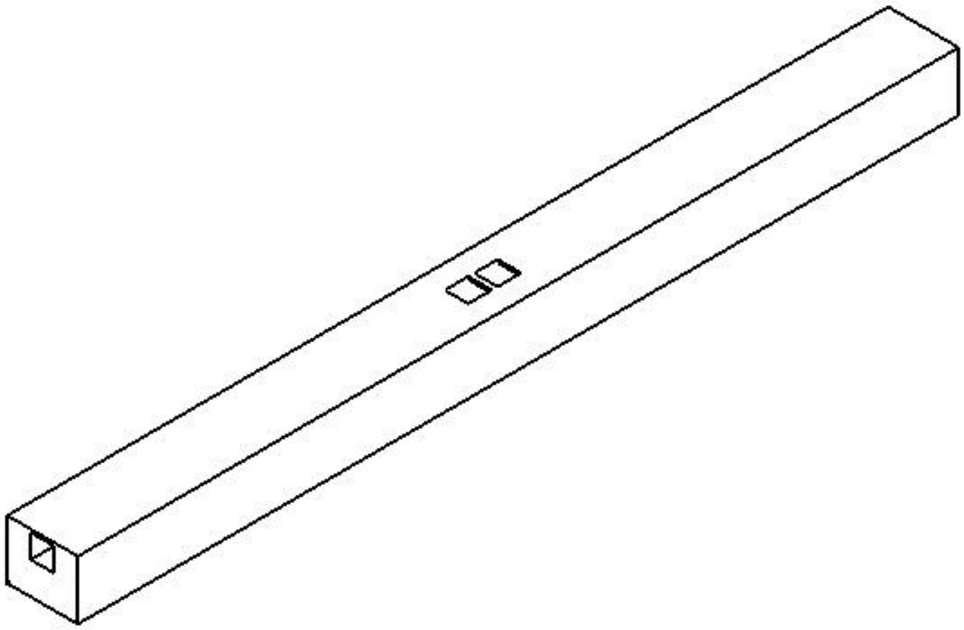

Fig 3.1 Isometric view of the model

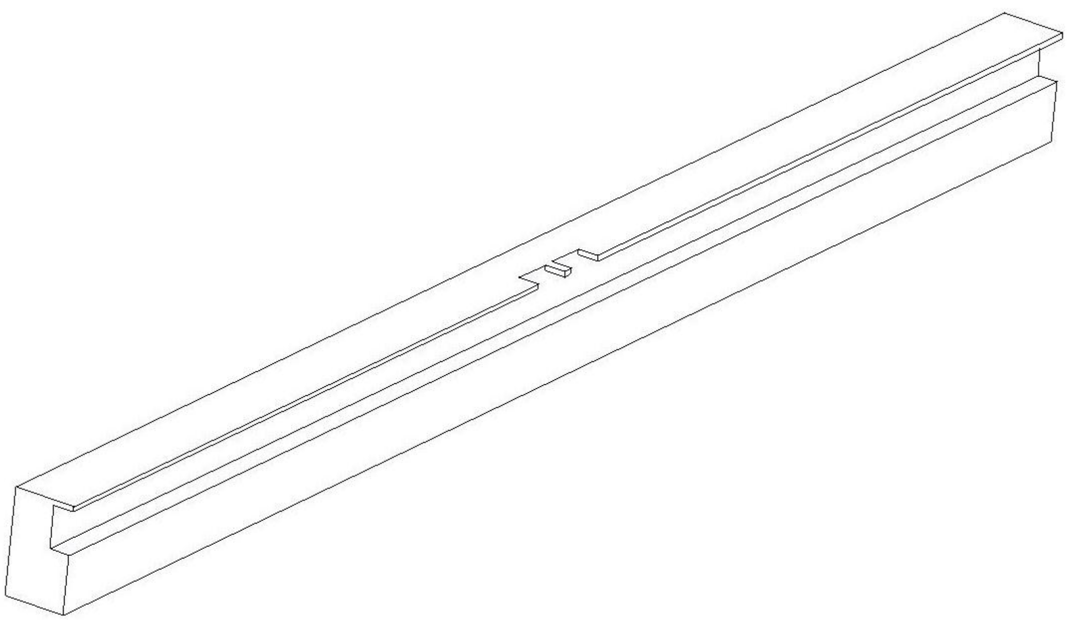

Fig 3.2 Computational view of the model

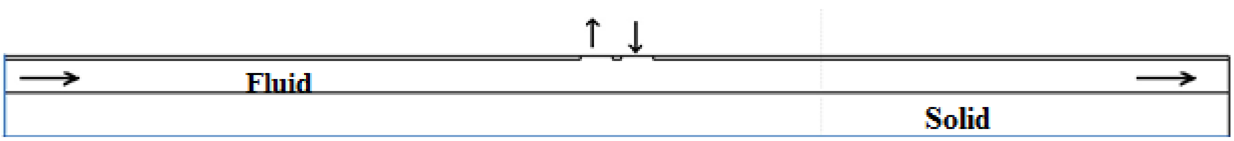

Fig 3.3 Recharging microchannel with fluid flow direction

3.5 Boundary Conditions:

(i) Constant heat flux on bottom wall (q''), W/m^2

(ii) Other walls are adiabatic i.e. dq/dx = 0, dq/dy = 0, dT/dx = 0 , dT/dy = 0

(iii) Velocity at inlet is constant, m/s

(iv) At outlets, pressure conditions are given

The governing equations are solved for the given boundary conditions using the FLUENT solver in ANSYS 15.0 . The flow is selected as laminar so no extra variables were needed unlike a turbulent model analysis. The SIMPLE algorithm is used to solve the equations. Under relaxation factors density and energy is taken as 1, pressure as 0.3 and momentum as 0.7. Convergence criterion for momentum and continuity is taken as 10^{-6} and for energy 10^{-8}. The meshes that were generated are taken in sizes of 0.045.

Due to use of academic license of ANSYS 15.0, simulation of meshes woth size less than 0.04 were not possible as it created more than 57,000 cells.

3.6 Grid Independence Test:

A grid independence test has been done to avoid any discrepancy in the values obtained due to grid size. Various grid sizes were taken on a simple channel and the results were plotted as in Fig 3.4. The size that showed the least discrepancy was chosen as mesh size for the rest of models. Meshing was done using multiphase meshing by setting relevance at 100, and the smoothing at high and relevance center to fine. The grid sizes taken were 0.05, 0.045. 0.04 (measure of the smallest edge). It has been found from the grid independence test . It was found that with mesh size 0.045, the results showed 0.008 % discrepancy with that of 0.05 and 0.0012 % with that of 0.04. So 0.045 was chosen as the grid indpendent mesh for the rest of the cases.

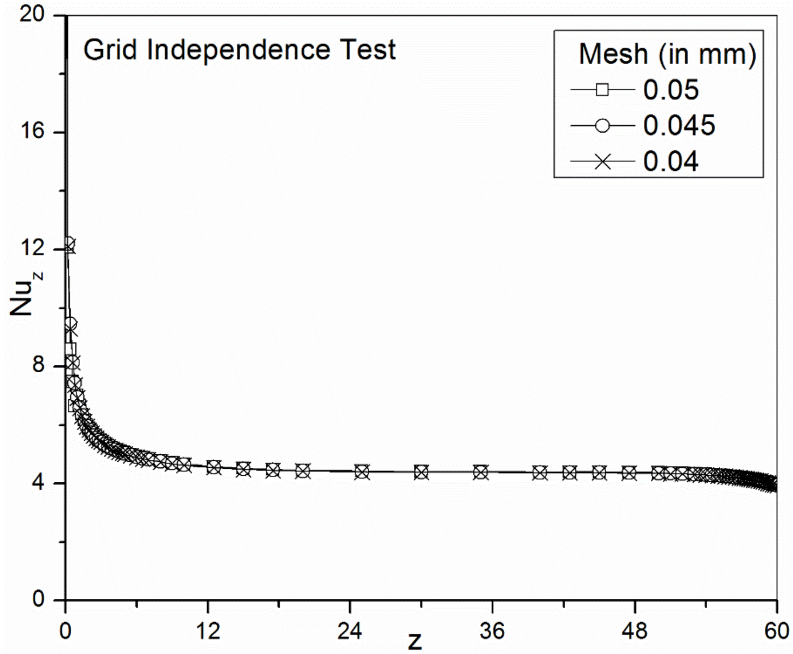

Fig. 3.4. Nusselt number (Nu_z) variation along the axial length (z) for three different size grids.

3.7 Data Reduction:

The analysis involved has too many variables included making the post processing work for graphs a bit tedious work. So the concerned quantities are changed to dimensionless quantities for easy plotting and observing of graph. The main quantities concerned are flux, temperature, axial distance, geometry, Nusselt number.

(a) Nusselt no is already a dimensionless quantity given by the formula $Nu_z = \frac{hD_h}{k_f}$, which needs no further reduction.

(b) The flux applied at the bottom face (q) and the flux experienced at the conjugate wall (q′) are different because of varying surface area. To study the variation of flux the term considered is ϕ, dimensionless flux defined as $\phi = \frac{q'}{q}$.

(c) The temperature variation is analysed by defining a dimensionless temperature represented by Θ, defined as $\Theta = \frac{T - T_i}{T_f - T_i}$. It can be defined both for fluid temperature and wall temperature, the input and output temperatures are that of fluid for both cases.

(d) The axial direction z is also made dimensionless by defining $Z^* = \frac{Z}{\mathrm{Re} \cdot \mathrm{Pr} \cdot D_h}$.

(e) The geometrical variation is defined by $\delta_{sf} = \delta_s / \delta_f$ where δ_s represents thickness of solid substrate and δ_f represents fluid cross-section width.

These parameters simplify the representation of the variables and easy study of the variations. These are therefore used throughout the thesis for representing the above mentioned quantities.

Chapter 4

Results and Discussion

The objective of this study is to analyze the thermo-hydrodynamic flow of a recharging rectangular microchannel and determine its heat transfer characteristics. Basically there exists two conditions for analyses of these type of flow: (a) at constant temperture, (b) at constant heat flux boundary condition at the outer surface of the channel. Constant temperature system are rarely encountered in practice where as constant heat flux is a common type of system for many cooling applications. So constant flux boundary condition on the bottom of the substrate is considered in this study.

Before moving on to 3-D modelling of the recharging concept, first the 2-D simulation is done. With the same length i.e. 60 mm and the Nusselt number variation so as to if it meets the expected results. For 2-D simulation, a fluid (water) flow between two plates separated by a distance of 0.4 mm, bottom plate being 0.4 mm thick, is subjected to a constant heat flux at its bottom, and the top surface of the upper plate (0.2 mm) is insulated. The recharging outlet is provided on top plate at 29.5 mm (0.4 mm width) and recharging inlet at 30.1 mm (0.4 mm width). The Reynold's number is taken as 100. Further calculations are made to get the desired boundary conditions . The velocity at inlet is found to be 0.2512 ms^{-1} corresponding to Re = 100. The flux provided is calculated to be 69909.1 W/m^2. Now to find the pressure to be applied at the recharging outlet. A simulation is done on a half channel (30 mm) and then the calculated pressure drop is applied at the outlet. This pressure was found to be 634 Pa. The resulting velocity profiled matched the required condition, and also the local Nusselt number varied exactly as presumed as can be seen in Fig. 4.1. This confirmed the viability of the recharging microchannel concept. The average Nusselt number increased owing to the increase in the developing region of the channel because of two developing zones for the same overall channel length. This result compelled a detailed analysis on the 3-D model. The 2-graphs were also compared with a split wall type pipe flow i.e. similar to pipe flow in two pipes in series. It is found that the recharging case is intermediate between those two cases.

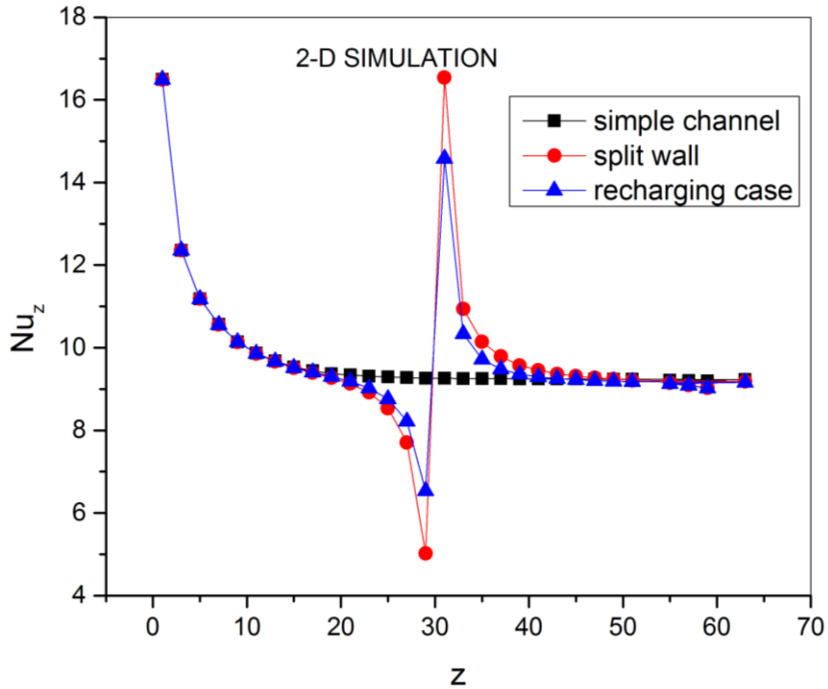

Fig. 4.1 Nusselt number (Nu_z) along the axis (z) for three different cases of 2D flow

Proceeding to the 3-D simulation, the geometry was modelled in solidworks and meshed according to grid indpendence test. There after the governing differential equations are solved using the boundary conditions. At first for Re = 100, solid substrate material (k_s) was varied for water as the working fluid which enters the system at 300 K. After the first simulation velocity contour at the recharging section is studied and ensured if the desired condition is achieved or not. The Pressure should be set such that there should be no backflow of water from recharging inlet to the recharging outlet because that would create a turbulence zone as shown in Fig. 4.2, in between these two sections disturbing the parametric variation and safe working of the section. The pressure should thus be calculated accordingly a bit accurately to avoid disputed calculation of the other flow parameters in microchannel.

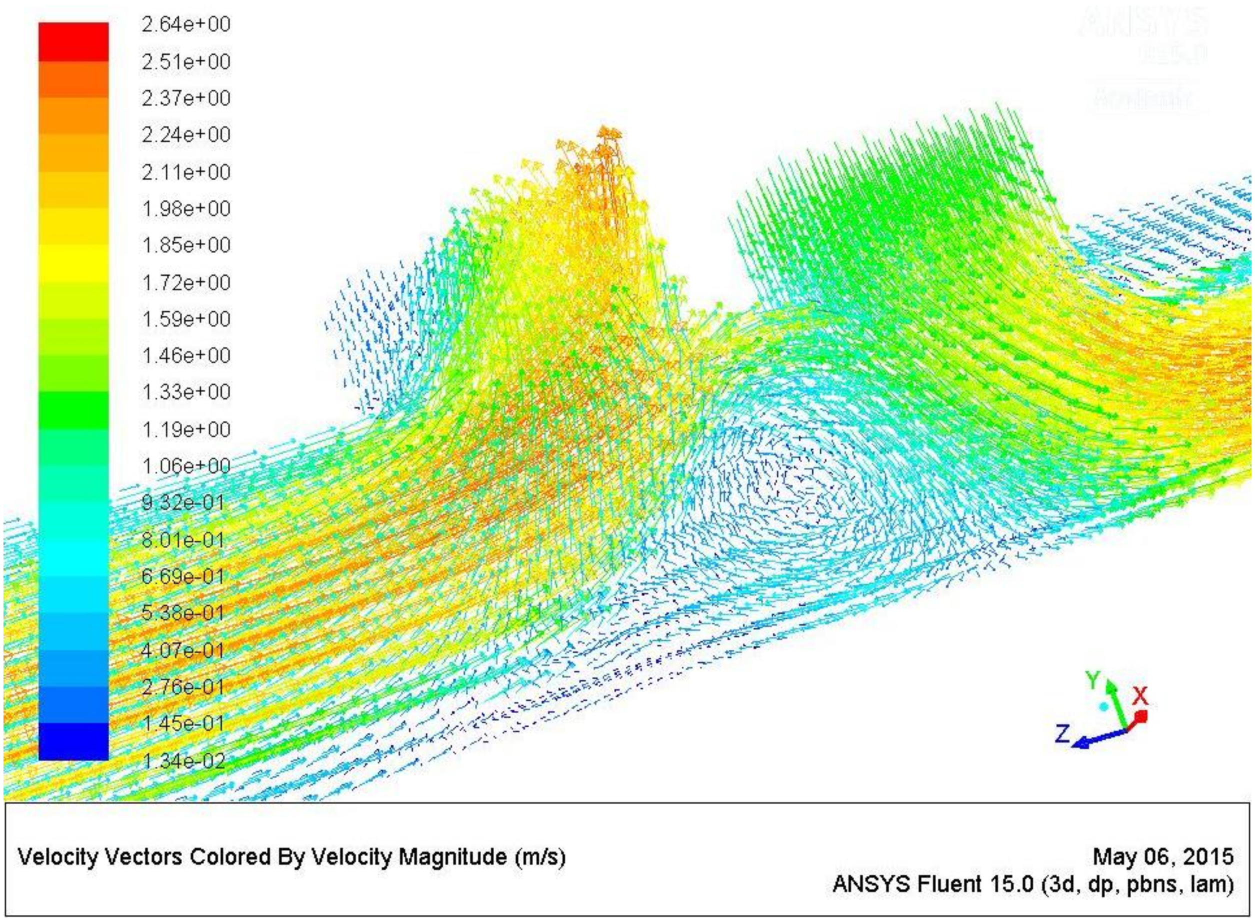

Fig. 4.2 The turbulence zone and backflow due to incorrect pressure condition at secondary outlet

After the post-processing of the data obtained, various plots are plotted to understand the variations in the microchannel flow. The recharging effect was commonly observed in all of the plots due to two developing zones. The variation of dimensionless flux, dimensionless temperature of wll and fluid and the local Nusselt number variation. The comparison of the parameters are done by plotting the parameters with varying z* and the variations are studied accordingly. For the comparison process, only the lowest,maximum and the k_{sf} corresponding to the maximum nusselt number (constantan~37.68) is used.Starting with the variation of the dimensionless flux as shown in Fig. 4.3.

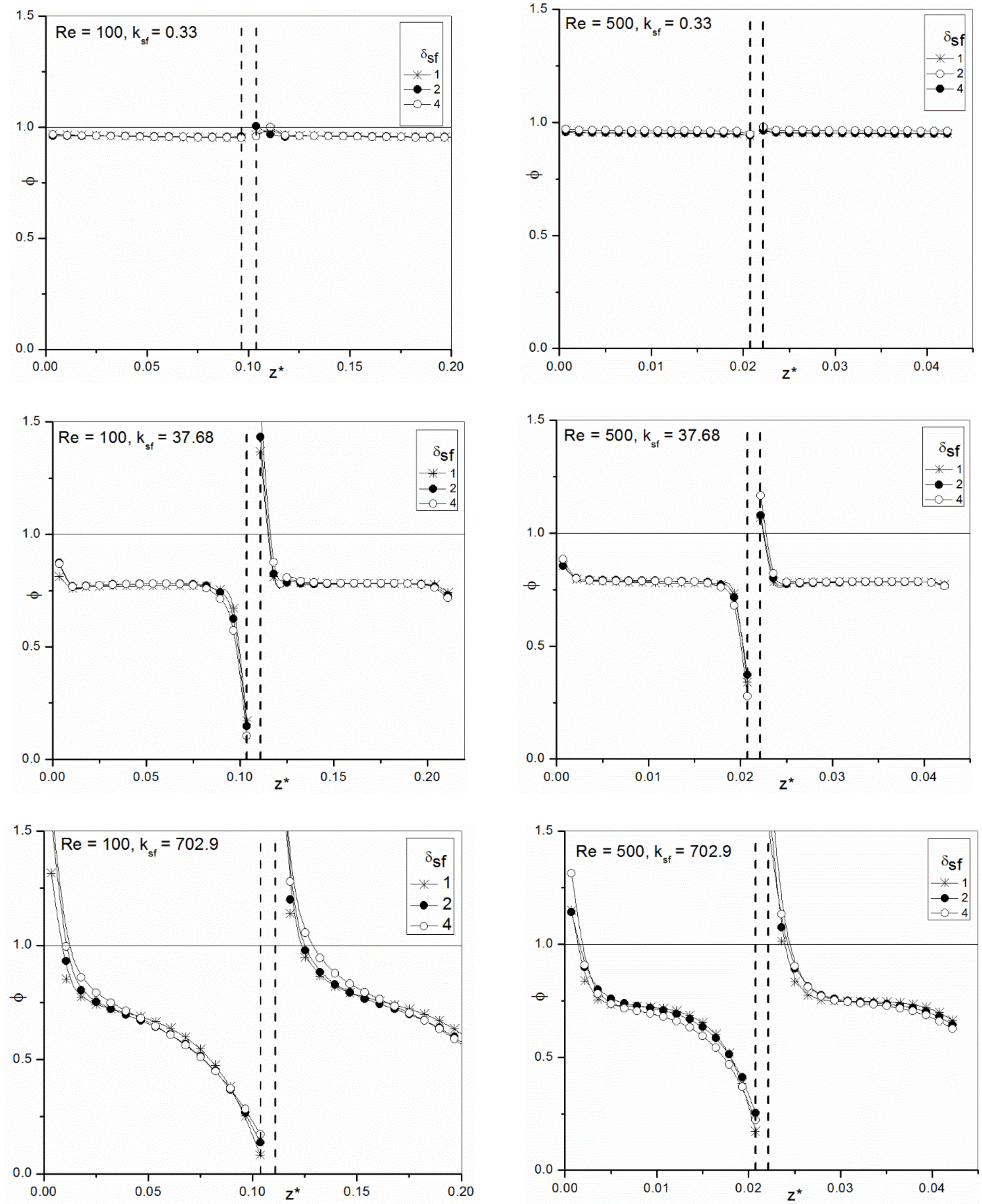

Fig. 4.3 Axial variation of dimensionless flux for number for range of simulation variables.

The heat transfer process from substrate to fluid channel is decided by boundary condition experienced at conjugate walls of the channel. The dimensionless flux ϕ is calculated for observing the variation of this flux. At low conductivity ratios, irrespective of δ values, actual heat flux experienced is nearly equal to the flux observed at the bottom surface. But at high conductivity the heat-flux observed at conjugate wall deviates from that of constant heat flux provided at bottom. This is mainly seen due to increase in axial back conduction due to decrease in thermal resistance of substrate, as the conductivity increases.Irrespective of the Reynold's number the pattern remains same for the flux variation.

The axial variation of wall and fluid temperature (dimensionless) as can be seen from the Fig. 4.5 is that at low k_{sf} the rise is as per the normal channels , doubling at the recharging section till the fully developed flow, after which it shows a constant temperature difference. But at higher thermal conductivities, the rise in fluid temperature is no more linear thus deviating from the "well-defined" condition of rise in temperature profiles. The wall temperature variation clearly indicates heat flow from the downstream location of the substrate toward the upstream direction. This tends to "isothermalize" the wall temperature on the fluid–solid boundary. As δ_{sf} increases the variation could be easily pointed out.

The dimensionless wall and fluid bulk temperature variation is as follows

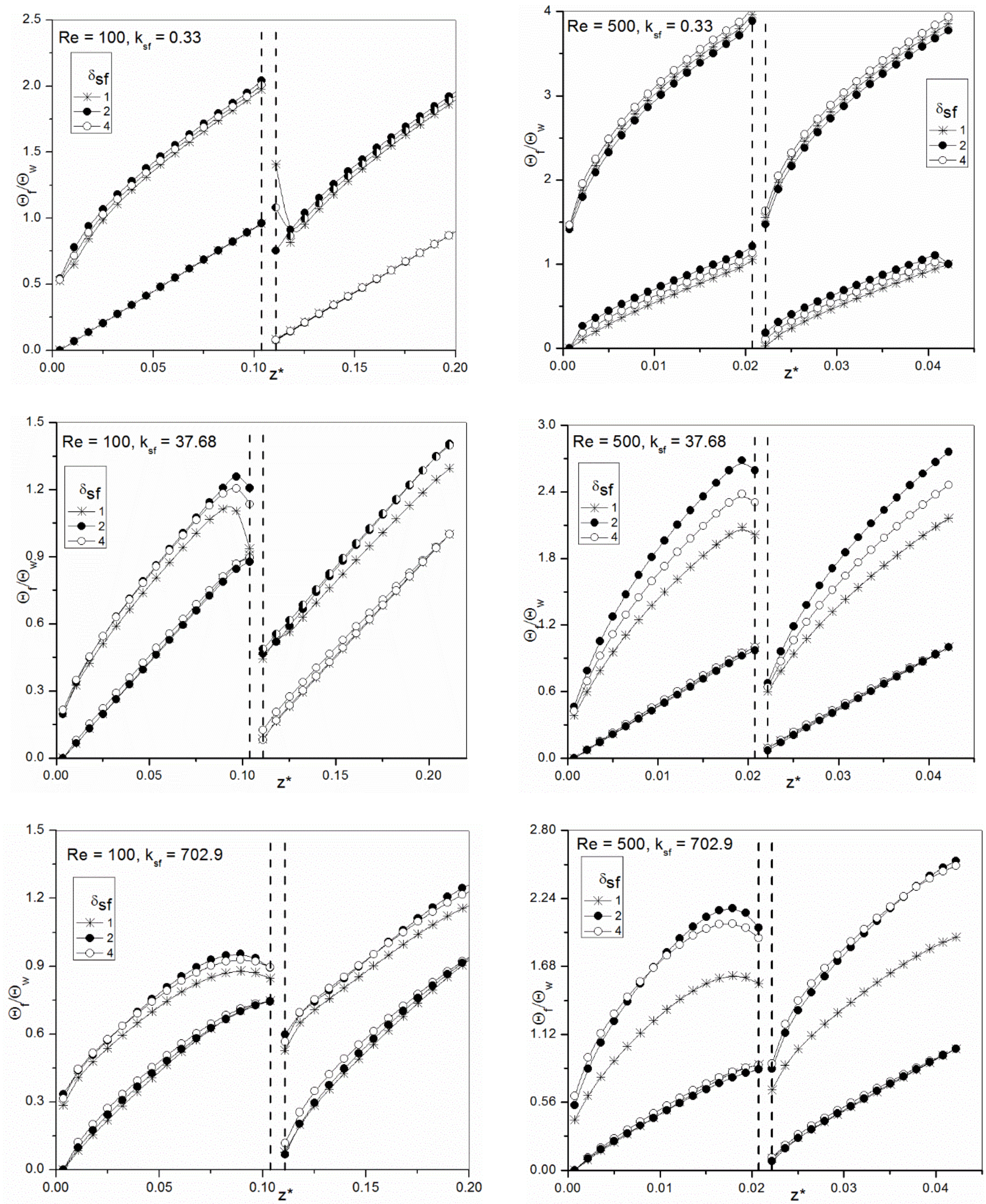

Fig. 4.4. Axial variation of dimensionless wall temperature and bulk fluid temperature for range of simulations

As can be seen in Fig. 4.5. The Nusselt number graph is spiked in middle in the recharging section that is the main cause of increased nusselt number than conventional microchannel. As it can be inferred from above figures it can be seen that the highest Nusselt number is seen at k_{sf} = 37.68, The lowest k_{sf} has the lowest value and highest k_{sf} has value near to the maximum but less than it. This validates the effect of thermal conductivity as discussed. At low thermal conductivity, the resistance is high so Nusselt number is less because of low vertical interface heat transfer whereas at very high thermal conductivity due to decrease in the thermal resistance, the axial back conduction increases thereby decreasing the Nusselt number. Also it can be inferred with increasing Reynold's number and higher δ, the average converges away from 3.55 i.e. most of value lies beyond 3.55 indicatin a pseudo-isothermal condition due to axial back conduction but at low k_{sf} values, values are in range of 2.72-3.55 indicating conduction only through bottom wall.

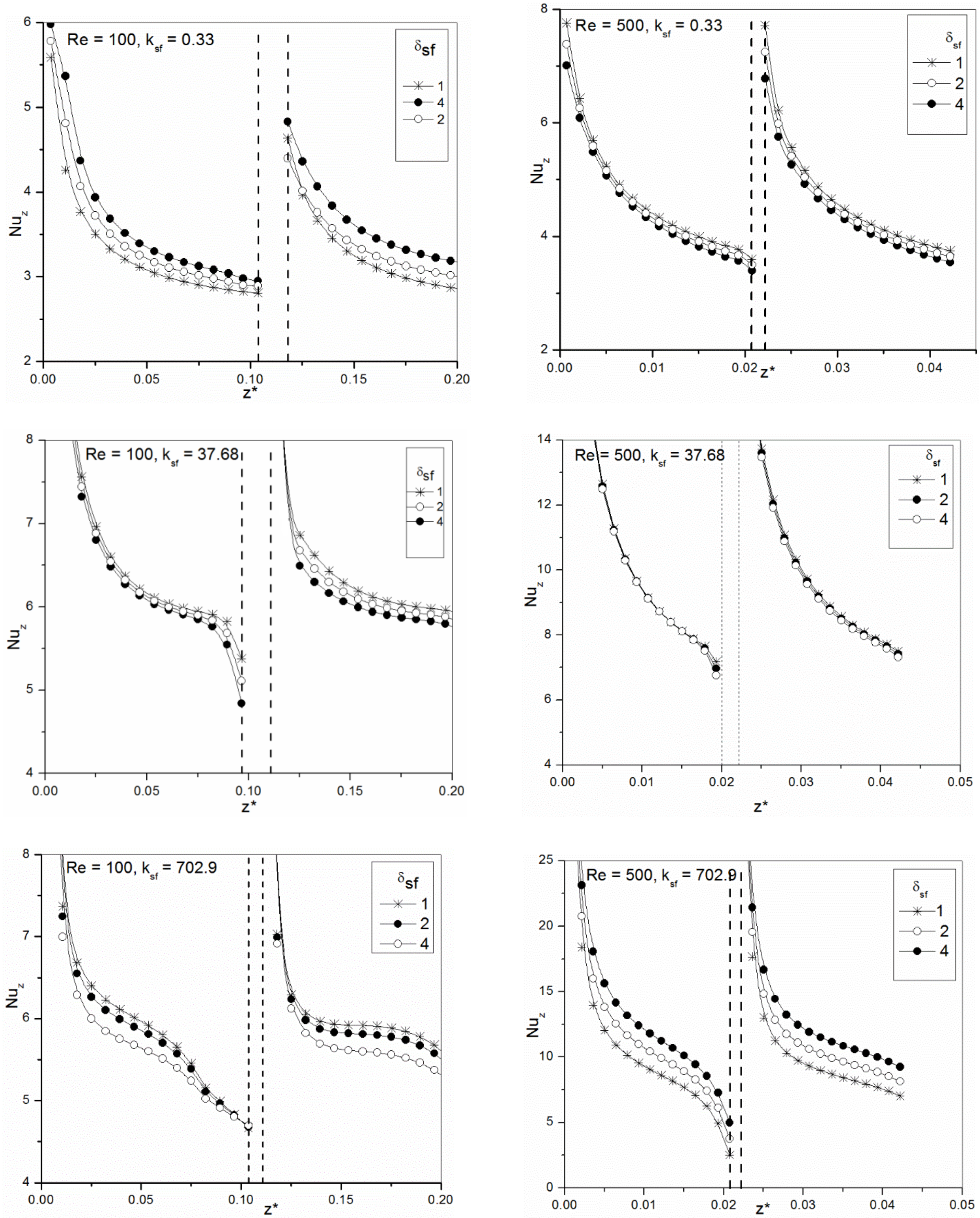

Fig 4.5 Axial variation of local Nusselt number for the range of simulations

The average Nusselt number was calculated for the cases and the data are processed to calculate Nu_z varying with k_{sf} values and the optimum k_{sf} was decided. For a wider parametric variation, in total 11 k_{sf} values (materials) are considered in the numerical study. But only the lowest, highest and optimum k_{sf} at which average Nu is maximum, is only used to compare the cases for different δ_{sf} values (varying substrate thickness (δ_s)). The boundary conditions remained same as the case of 2-D simulation. The pressure to be applied at recharging outlet is found by the same method as before and it was found to be 3609.19 Pa. The same graph was also plotted for the first geometry by varying Reynold's number to 500. For Reynold's number 500, the velocity changed to 1.256 m/s , heat flux changed to 354595 W/m^2 and the pressure was changed to 15630 Pa. The resulting plot for δ_{sf} = 1 is presented in Fig. 4.6.

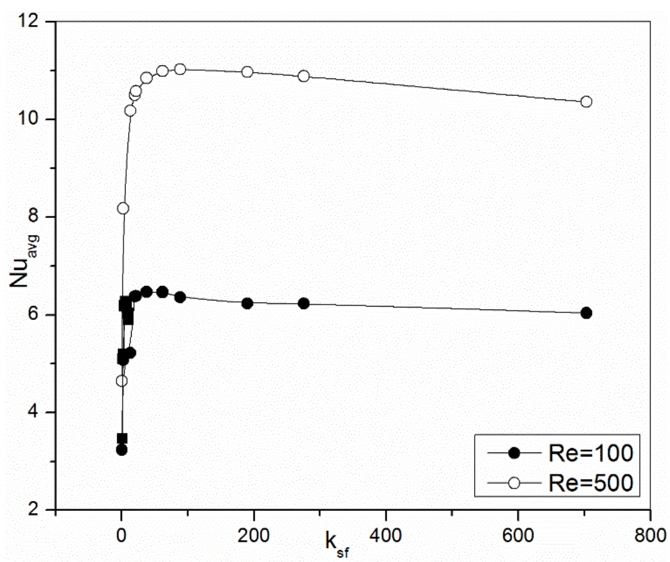

Fig. 4.6 The variation of Nu_{avg} with varying k_{sf} values for Re = 100, 500 and δ_{sf} = 1.

As it can be inferred from above, the maximum Nusselt number is observed around the thermal conductivity values of 40-55. The closest value we have to these range corresponds to the material constantan with k_{sf} value of 37.67~38 . Thus this is taken as the optimum k_{sf} for maximising Nusselt number. As it can be seen from Fig. 4.4 for various deltas also the variation of the above graph remains nearly same . The Nusselt number spikingly increases for small k_{sf} values just near to 0 due to commencemnet of heat transfer till it reaches a maximum value. After that the Nusselt number again starts decreasing with increasing k_{sf} values with a moderate slope which is explained due to increasing conjugate effect. Also very high thermal conductivity values, decreases the thermal resistance considerably thereby increase axial back flow of heat thus decreasing Nusselt no. At a given k_{sf} the main factor is Reynold's number rather than geometry in deciding a maximising value which is a very useful inference.

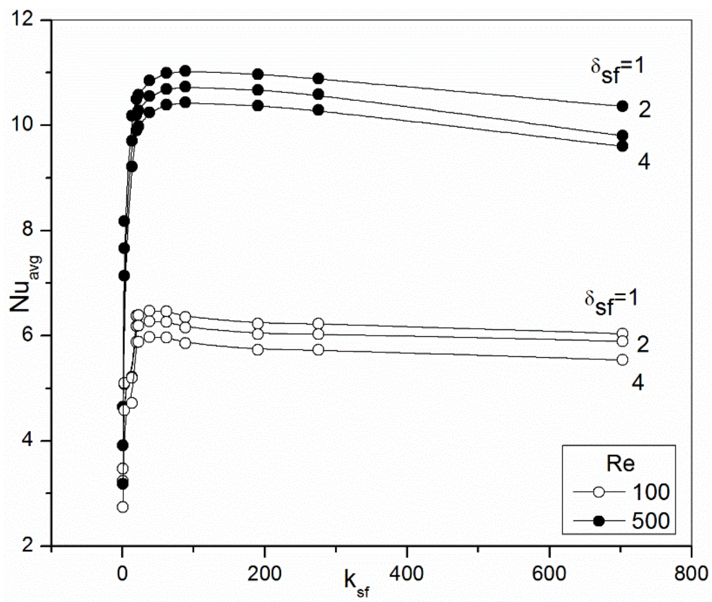

Fig. 4.7 Nu_{avg} varying with k_{sf} for different values of Re and δ_{sf}.

Thus it is clearly observed k_{sf} plays the major role in the variation of Nusselt number rather than the geometry i.e. the thickness of solid substrate. In view of the above results it can be concluded for the given range of simulations , the maximum Nusselt number is obtained for δ_{sf} = 1, for Reynold's number Re = 500 and the optimum k_{sf} value for k_{sf} = 37.68 (Material: constantan). The Nusselt number is compared for this case of recharging and simple microchannel and the resulting graph is shown in Fig. 4.8.

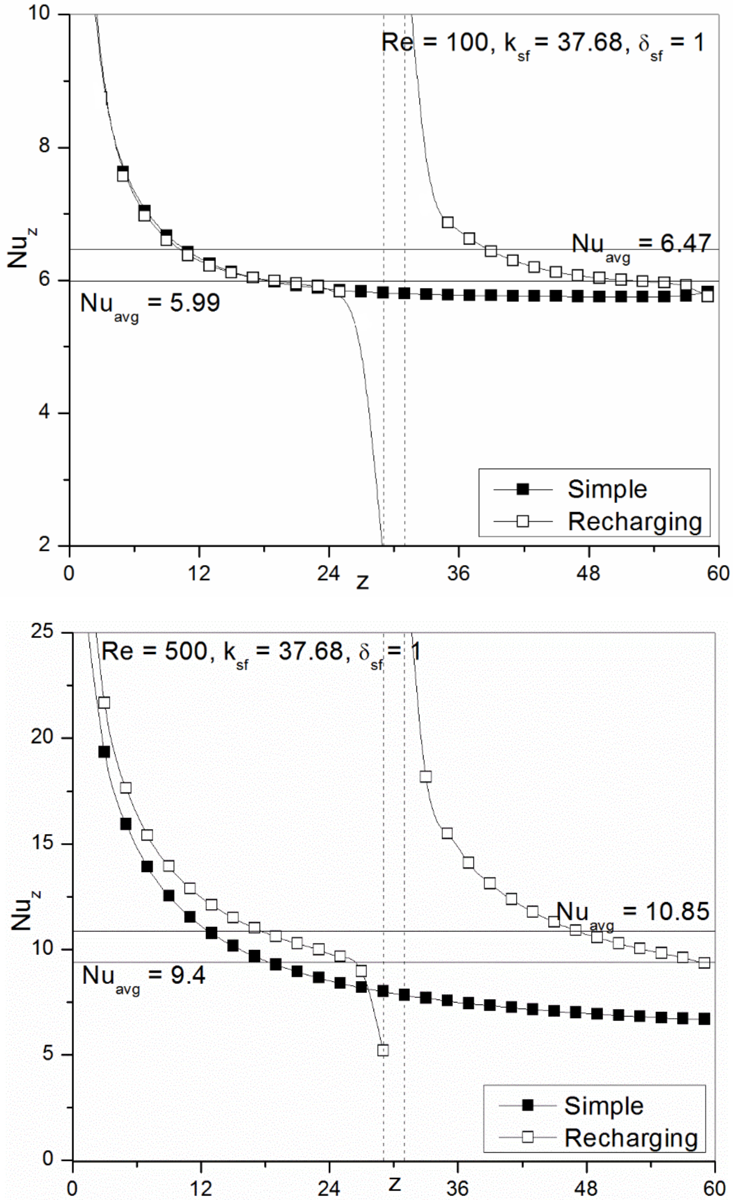

Fig. 4.8 Comparison of average Nusselt number in simple as well as recharging microchannel for (a) Re =100, k_{sf} = 37.68, δ_{sf} = 1 and (b) Re = 500, k_{sf} = 37.68, δ_{sf} = 1.

As it can be seen that due to recharging of microchannel, the average Nusselt number increases for Re = 100, by around 8% whereas for Re = 500 by around 15 % for the above case. Similarly the other cases also see various increment in the Nusselt number due to recharging.

Chapter-5
Conclusion

5.1 Inference from the results:

For a wider parametric variation for better constrast of the outcome of the observation based on the numerical study under taken, in total 66 cases are considered (Re = 100, 500, δ_{sf} = 1, 2, 4, k_{sf} = 0.33 – 702.9). From the above range of simulations and observations various inferences could be taken in account of the proposed recharging microchannel concept. The notable outcome of the observations are as follows:

(a) Recharging of microchannel "theoretically" increases the average Nusselt number as can be seen in previous chapter for the maximum case at Re = 500, k_{sf} = 37.68 and δ_{sf} = 1 an increase of about 15% is noted by the introduction of recharging channel, similarly other cases also shows a positive increment in the Nusselt number but its practical viability is yet to be ventured upon .

(b) Thermal conductivity ratio k_{sf} plays an importnt role in maximising microchannel as its effect on the nusselt number is more than any other factor clearly visualised from graph variations.

(c) The axial back conduction plays a major role in affecting microchannel heat transfer and thus should be studied on carefully, the wall temperature and fluxes are mainly affected due to this variation.

(d) Maximising Nusselt number is achieved at optimum conductivity ratio determined by studying the variation of Nusselt number throughout the range of conductivities available.

5.2 Scope for future work:

Further venturing into this concept, we could calculate the combined effect with introduction of more than one recharging area in the conduit. Also a more wide range of simulation could be considered for a more clear picture of the phenomenon. Added to it for validating a practical drafting of the idea, the portion between the recharging outlet and inlet has to be studied carefully to check the possible problems created by the vaccum created or by the backflow of the fluid. This idea holds a good amount of potential to enhance the efficiency of microchannels and therby spread its range of application to a wider area.

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